IN-LINE FORCES ACTING ON SMOOTH CYLINDERS IN HARMONIC FLOW

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THESIS

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IN HARMONIC FLOW

by

Savas Onur

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The drag and inertia coefficients have been determined through the use of the Fourier analysis and found to depend on Reynolds number as well as Keulegan-Carpenter number for Reynolds numbers

about greater than 25,000. A new parameter namely the "Frequency Parameter" has been defined as the ratio of the Reynolds number to the Keulegan-Carpenter number, and the existence of unique relations between the frequency parameter and the drag and inertia coefficients have been shown.

It is recommended that the experiments be extended to even larger Reynolds numbers using the same experimental technique and apparatus.

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in Harmonic Flow

by

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NOMENCLATURE

A	Virtual amplitude of the motion
A_1	Amplitude of the motion
a_m	Maximum acceleration
C_D	Average drag coefficient
C_M	Average inertia coefficient
C_{FMAX}	Maximum in-line force coefficient
D	Diameter of the test cylinder
F	Instantaneous total force acting on the test cylinder
F_D	Drag force acting on the test cylinder
F_I	Inertia force acting on the test cylinder
F_m	Measured force
g	Gravitational acceleration
H	Distance between pressure taps and mean water level
K	Keulegan-Carpenter number
L	Length of the test cylinder
Re	Reynolds number
s	Distance between the pressure taps
T	Period of oscillations
t	Time
U_m	Maximum velocity
u	Instantaneous velocity
w	Width of the test section
β	Frequency parameter
γ	Specific weight of fluid

Δ	Differential pressure
θ	Phase angle
ν	Fluid kinematic viscosity
ρ	Fluid density

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I. INTRODUCTION

Information about the time-dependent forces acting on bluff bodies in general and circular cylinders in particular has considerable practical interest in ocean and wind engineering, as well as in basic understanding of fluid mechanics. Extensive discussion about the flow-induced forces and oscillations exists in the literature, but the basic hydrodynamic data are lacking, particularly at higher Reynolds numbers which are of current practical interest.

Much of the present knowledge on the hydrodynamics of the oscillatory flow about the cylinders has been obtained by means of model tests in wave channels at Reynolds numbers generally two to three orders of magnitude smaller than prototype Reynolds numbers. These model tests, which have been primarily concerned with in-line forces, have disclosed that the drag and inertia coefficients may be correlated with the amplitude of the motion relative to a characteristic dimension of the body and that the effect of the Reynolds number is obscured by that correlation. The field data have shown rather large scatter and no systematic correlation with either the relative amplitude or the Reynolds number. Understandably, there is considerable controversy in the literature regarding the effect of viscosity, free surface, etc.

In view of the foregoing considerations, the present research program was undertaken with the following objectives:

(a) To furnish data, obtained under carefully controlled laboratory conditions, about the in-line and transverse forces acting on smooth cylinders in a harmonically oscillating fluid at Reynolds numbers up to 400,000 and

(b) To identify the physical mechanism and parameters responsible for the correlation or scatter of the force-transfer coefficients.

This thesis deals only with the in-line forces acting on smooth circular cylinders. It does not deal with ocean waves, non-harmonic fluid oscillations, wave and current combinations and its consequences, diffraction effects, free-surface and/or wall proximity effects, fluid elasticity or hydro-elasticity of flexible or flexibly supported cylinders in harmonic motion, or finally the interference effects between neighboring structural elements. Furthermore, no attempt is made to offer a chronological and/or critical survey of the fluid loading on cylinders or offshore structures. Fairly complete accounts in the context of wave forces are given by Wiegel (1), Hogban (2), and Grace (3), where an extensive list of references can be found. Only those works which have a direct bearing on the evolution and discussion of the present data will be reviewed, in some detail, wherever appropriate.

In closing the introduction it should be emphasized that the force-transfer coefficients must be obtained experimentally for the analysis of the harmonic flow about cylinders may never be forthcoming.

II. ANALYSIS OF THE DATA AND THE FORCE COEFFICIENTS

A completely satisfactory analysis of the resistance in unsteady separated flow has escaped the concentrated efforts of many researchers. No theoretical model can, at present, predict the complete force and flow characteristics of a harmonic motion about a circular cylinder. In the absence of such an analysis, the most serious difficulty lies in the description of the time-dependent force. Other difficulties arise in the description and interpretation of the history of the flow and the effect of the vortices and turbulence in the flow approaching the body at a particular time in the oscillation. One approximate and physically meaningful way around these difficulties is to assume, following Stokes' classic analysis of the oscillating pendulum, that the total time-dependent in-line force may be expressed as a sum of the velocity-squared dependent drag and the acceleration dependent inertial force, each with a suitably averaged force coefficient. This then is the basis of the so-called Morison's (4) equation. For a circular cylinder it may be written as

$$F = F_D + F_I = 0.5\ C_D\ L\ D\ \rho|U|U + 0.25\ \pi\ \rho\ L\ D^2 C_M \frac{dU}{dt} \qquad (1)$$

For an oscillating flow represented by $U = -U_m \mathrm{Cos}\theta$, with $\theta = 2\pi t/T$, the Fourier averages of C_D and C_M are given by Keulegan and Carpenter (5) as

$$C_D = -\frac{3}{4} \int_0^{2\pi} \frac{F_m \cos\theta}{\rho U_m^2 L D} d\theta \tag{2}$$

$$C_M = \frac{2 U_m T}{\pi^3 D} \int_0^{2\pi} \frac{F_m \sin\theta}{\rho U_m^2 L D} d\theta \tag{3}$$

in which F_m represents the measured force, L and D the length and the diameter of the cylinder, U_m the maximum velocity and, T the period of the flow oscillations. Evidently, C_D and C_M represent the first two terms in a complete series expansion of the normalized force in terms of the odd integers in sines and cosines. Previous studies by Keulegan and Carpenter (5) and Sarpkaya and Tuter (6) have shown that C_D and C_M as given by equations (2) and (3) are the most significant ones and sufficient to represent the measured force adequately.

A. GOVERNING PARAMETERS

The coefficients cited above will have to be correlated through the use of suitable parameters in order to show that they have some degree of universality within the range of the parameters encountered. A simple dimensional analysis of the flow under consideration (uniform harmonic motion about a circular cylinder placed with axis normal to the flow) shows that the time dependent force coefficient may be written as:

$$\frac{F}{0.5 \rho U_m^2 D L} = f\,(U_m T/D,\ U_m D/\nu,\ t/T) \tag{4}$$

$$= f\,(K,\ Re,\ t/T) \tag{5}$$

Evidently, $U_m T/D$ may be replaced by $2\pi A/D$ or simply by A/D where A represents the amplitude of the oscillations.

There is no simple way to deal with equation (5) even for the most manageable time-dependent flows. The evaluation of the instantaneous values of the force coefficients is not at present feasible. Another and perhaps the only alternative is to eliminate time as an independent variable in equation (5) and consider suitable time-invariant averages of the force coefficients. Thus, one has

$$\begin{bmatrix} C_D \\ C_M \\ C_{FMAX} \end{bmatrix} = f_i \; (U_m T/D, \; Re) \qquad (6)$$

Even this equation (6), as simple and as idealized as it is, gives rise to many questions: Do the averaged force coefficients really depend on both K and Re ; can one obtain meaningful conclusions by plotting the data for a given coefficient with respect to, say, K and connecting points having equal Re or vice versa ; how should the experiments be conducted so that equation (6) yields manageable plots ; which of the two parameters has a more pronounced effect on the force coefficient under consideration ; why has there been considerable scatter (see, e.g., Wiegel (1)) in the field data when plotted with respect to either K or Re ; are there ranges of K and Re in which the effect of one is obscured by a reasonable correlation of the force coefficients with the other? These and similar questions have been raised

by many other investigators and attempts were made to establish suitable correlations. The state of the art is such that the past conclusions and conjectures can be critically scrutinized only through the acquisition of data of high intrinsic quality obtained under controlled laborabory conditions with relatively simple harmonic flow situations. The purpose of such an effort is by no means to remove the need for actual full scale experiments. In fact it is to encourage full scale experiments and to enable those concerned to interpret and better understand the factors effecting the force-transfer functions.

Past experience (5,6) has shown that the force coefficients obtained under controlled laboratory conditions are primarily functions of K at relatively small Reynolds numbers ($Re < 25{,}000$) and that the effect of viscosity is obscured by the fairly good correlation between K and the force coefficients. Again, previous efforts and the reasoning based on dimensional analysis have shown that there is an undeniable effect of the Reynolds number. Thus, means have to be devised to delineate the effect of both K and Re or some other viscosity dependent parameter.

It is apparent that the maximum velocity U_m appears in both $K = U_m T/D$ and $Re = U_m D/\nu$. Simple rules of dimensional analysis state that one obtains the maximum amount of experimental control over the dimensionless variables, if the original variables, that can be regulated, each occur in only one dimensionless product. Thus, if U_m is easily varied experimentally, then U_m should occur in only one of the

independent dimensionless parameters. With this hint in mind, Re in equation (6) is replaced by $Re/K = D^2/\nu T$. This parameter shall be called the "Frequency Parameter" and denoted by β , so that:

$$\beta = \frac{D^2}{\nu T} \tag{7}$$

Evidently, for a series of experiments conducted with a cylinder of a given diameter D in water (of uniform and constant temperature and density) undergoing harmonic oscillations with a constant period of T , β is held constant, then the variation of a force coefficient with K may be plotted for constant values of β . Subsequently, one can easily recover the Reynolds number from

$$Re = \beta K \tag{8}$$

and connect the points, on each β = constant curve representing a given Reynolds number for a family of suitably selected values of the Reynolds number. Such a procedure eliminates the difficulty of trying to draw contours of constant K , or constant Re , or constant C_D or C_M versus K or Re , or K versus Re . Suffice it to note that the data reported here-in shall be analyzed according to the relationship

$$\underset{\text{a coefficient}}{C_i} = f_i\ (K,\beta) \tag{9}$$

and the Reynolds number will be used in the manner cited above. The power of this new procedure (new, as far as the wave force analysis is concerned) will become apparent later.

III. EXPERIMENTAL EQUIPMENT AND PROCEDURES

A. U-SHAPED OSCILLATING-FLOW TUNNEL

Experiments carried out by Sarpkaya and Tuter [6] in the initial phases of the study with small smooth cylinders at relatively low Reynolds numbers have proved the versatility and the usefulness of a U-shaped oscillating-flow apparatus. Thus, in an attempt to achieve larger Reynolds numbers, it was only natural to construct a larger U-shaped tunnel.

Among the various designs made by Professor Sarpkaya, the one shown in figure 1 was finally selected for construction. A photograph of the completed and fully instrumented structure is shown in figure 2. It consists of nine modules for ease of construction, transportation, and mounting. Each module is made of 0.95 cm. (3/8 inch) aluminum plates and reinforced 1.27 x 10 x 46 cm. (1/2 x 4 x 18 inches) aluminum flanges welded to the plates. The modules were assembled with the help of an air drying silicon rubber seal between the flanges of two adjacent modules and 2.54 cm. (1 inch) steel bolts placed 15 cm. (6 inches) apart. The inside of each module was precision machined so that the largest misalignment was about 1 mm. (0.04 inches).

Prior to the description of its instrumentation and operation, a few words are necessary about the general shape of the tunnel. The cross-section of the two legs is 183 x 91.5 cm. (6 x 3 feet). This selection was dictated

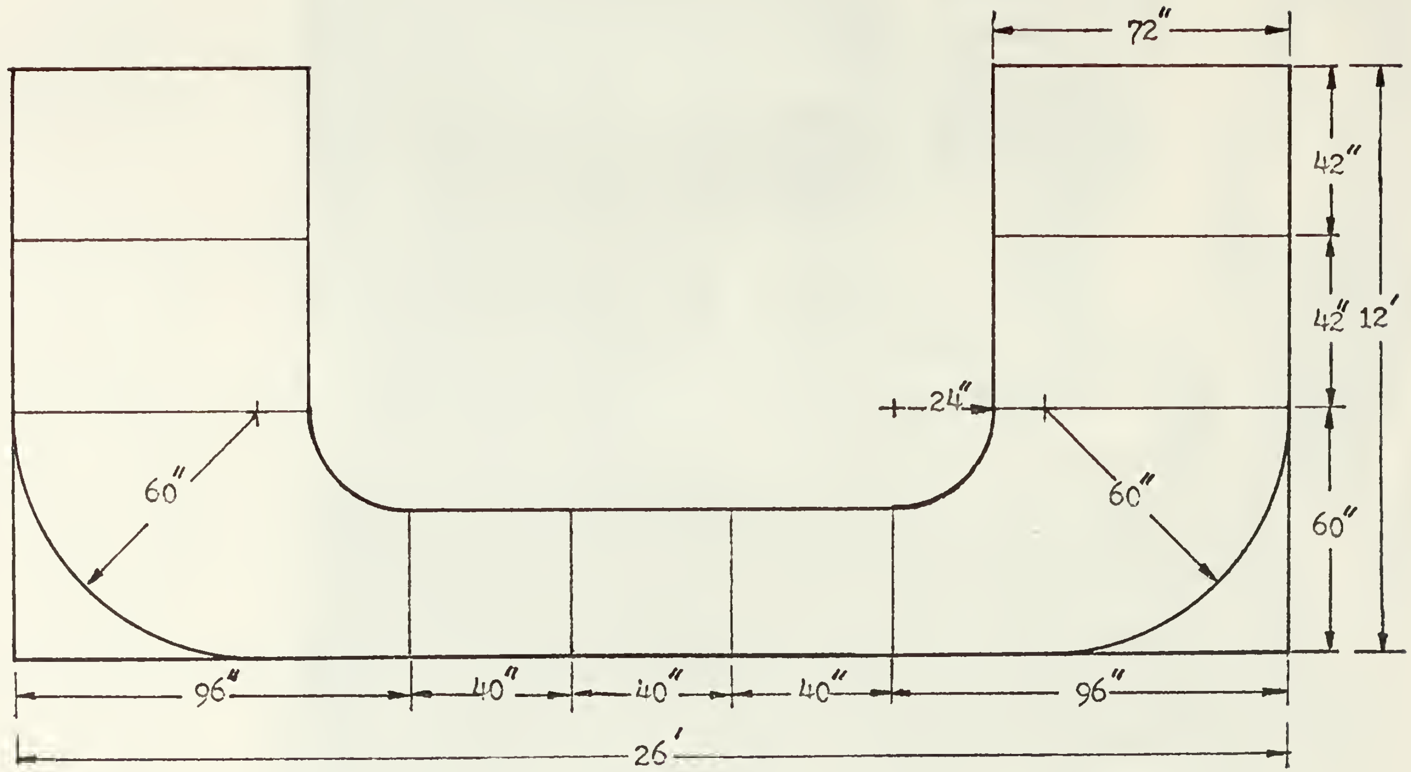

Figure 1. Schematic Drawing of the U-Tunnel

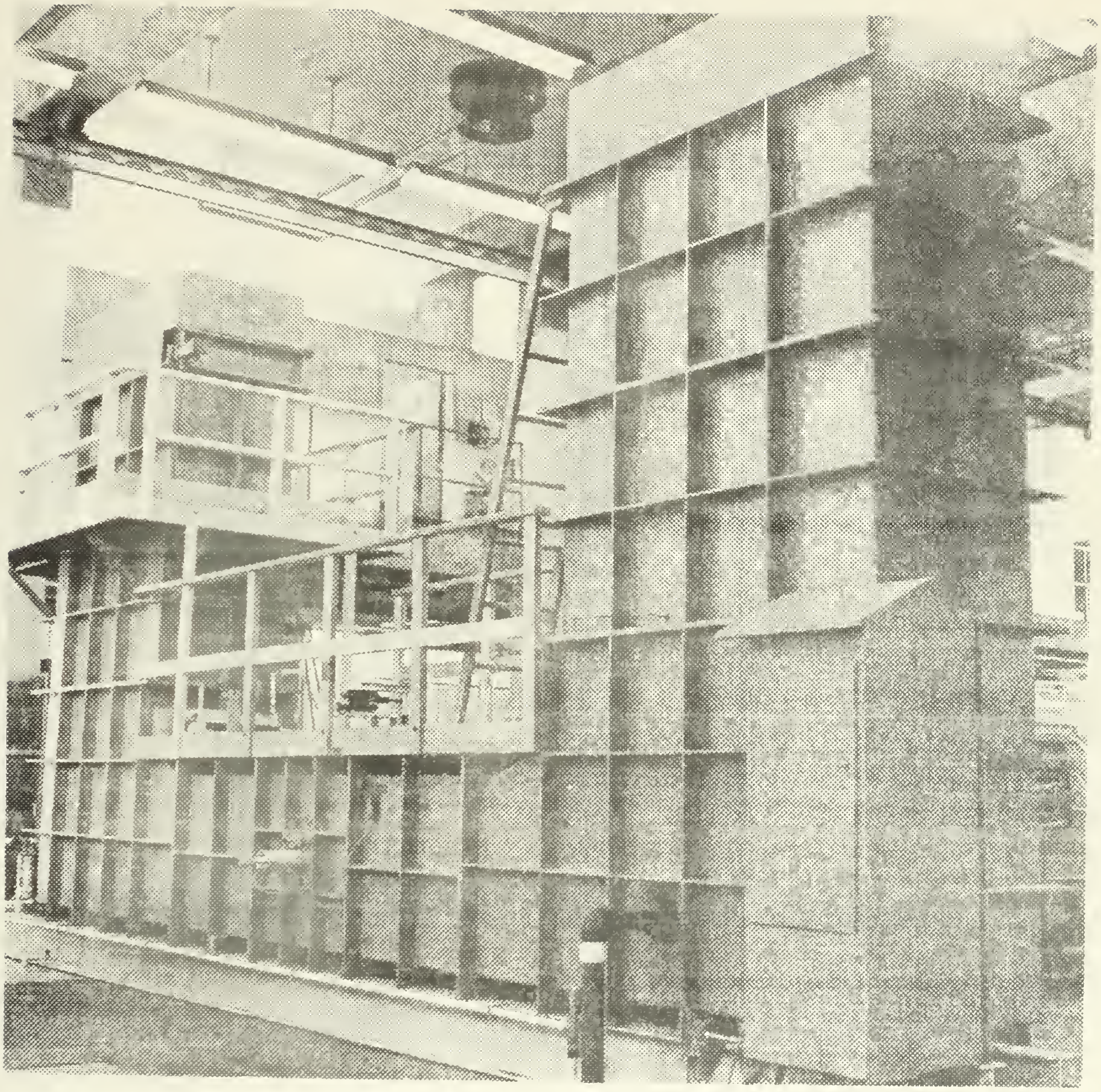

Figure 2. U-Tunnel

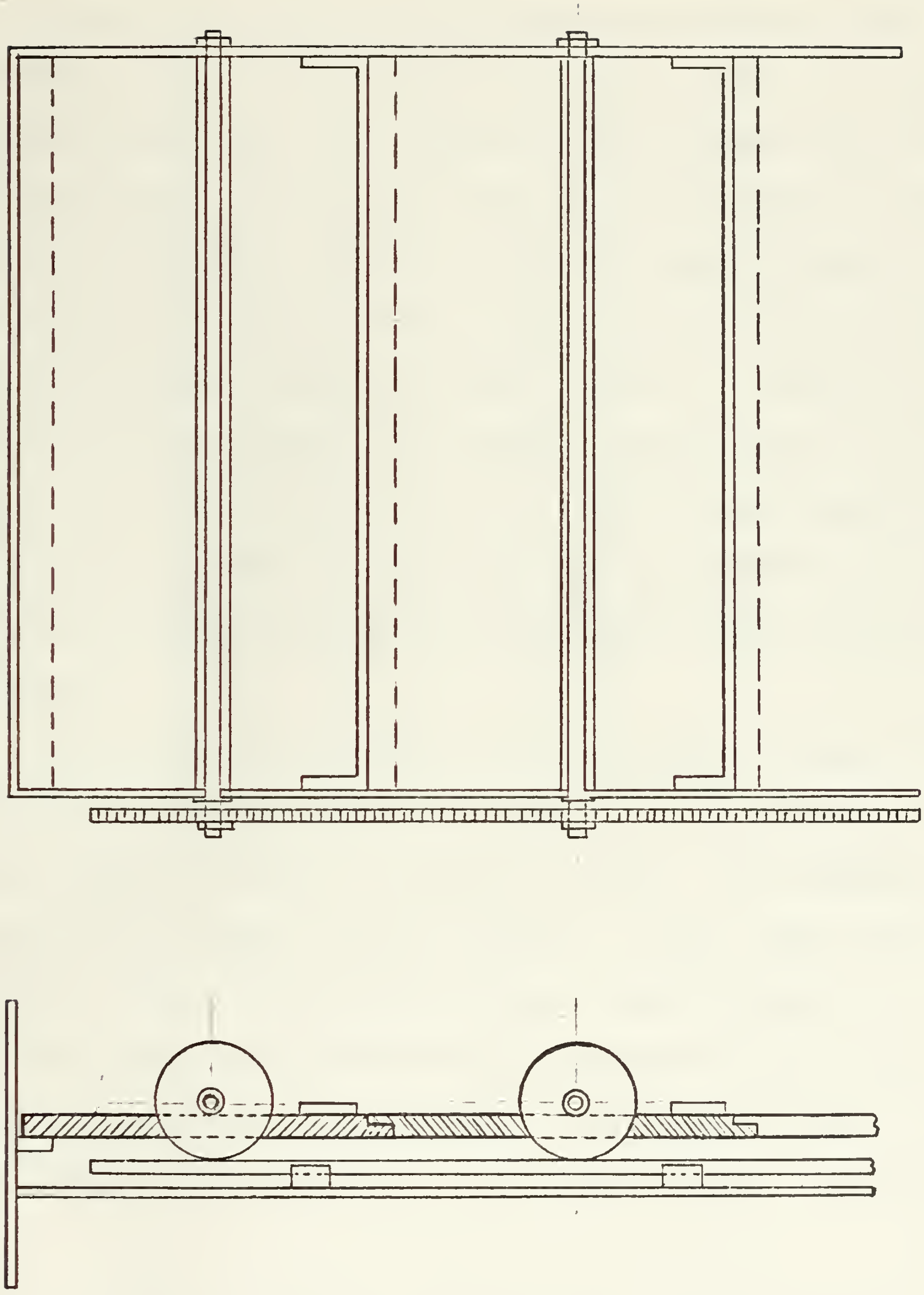

Figure 3. Schematic Drawing of the Butterfly Valve System

by several considerations such as available ceiling height, pressures to be encountered and hence the structural and economic considerations, desire to obtain a virtual amplitude* or velocity of oscillation at least twice that of the free surface, period of oscillation, Reynolds number and the relative amplitude A/D desired, natural-damping of the oscillations, and the magnitude and the frequency of the forces. The length of the horizontal test section was chosen larger than twice the virtual amplitude to insure fully developed uniform flow at the test section. Finally the two corners of the tunnel were carefully streamlined to prevent flow separation (see figure 1). This design proved to be more than adequate for no separation was encountered, and also the desired frequency and amplitude of oscillation were achieved.

The auxiliary components of the tunnel consisted of plumbing for the filling and emptying of the tunnel, butterfly-valve system, and the air supply system. The plumbing consisted of simple piping, valves, a small pump, and a filter. None of these will be described further.

The butterfly-valve system (mounted on top of one of the legs of the tunnel) consisted of four plates, each 4.6 cm. (18 inches) wide and 92 cm. (36 inches) long. A 2.54 cm.

*The virtual amplitude is defined as the amplitude of oscillation which the cylinder-experiences. Here it is exactly twice the amplitude of oscillation of the free surface in one leg of the tunnel.

(1 inch) steel shaft was placed at the axis of each valve plate. (see figure 3) Aluminum housings supported both ends of the shaft with self-aligning ball bearings. A 15 cm. (6 inches) gear was attached to one end of each shaft which extended beyond the bearing. All four valve plates were then aligned and driven by a simple rack and pinion system. The rack was actuated by an air-driven piston with the help of a three-way valve connected to the laboratory air supply system.

The valves, in their closed position, completely sealed the top of one of the legs of the tunnel. The top of the other leg was left open. Initially, the butterfly valves were closed and air was admitted to that side of the tunnel to create the desired differential water level between the two legs of the tunnel. Then the valves were opened quickly with the help of a pneumatically-driven three-way control valve. This action set the fluid in the tunnel in oscillatory motion with a natural period of $T = 5.272$ seconds. A series of experiments was conducted with one of the test cylinders to evaluate some of the experimental characteristics of the tunnel. It was found that the damping of the motion is such that the amplitude of oscillations decreases about 3 mm. (0.12 inches) per cycle and about 0.15 mm. (0.06 inches) per cycle for amplitudes half or less than of the maximum. Thus, over a period of four or five complete cycles of oscillation at any amplitude, the amplitude, velocity, and the acceleration of the fluid changed about one percent. Evidently the forced oscillations of the fluid, if such a method were to

be employed, cannot yield the amplitude to an accuracy better than one percent. Additionally in such a method one has to contend with some high frequency vibrations, however small, superimposed on the acceleration. These result from the cyclic operation of the butterfly valves. It is because of these considerations that the experiments were carried out by letting the system damp out the amplitude over many cycles of oscillations. The advantages of the method adopted become apparent very quickly. Firstly, the oscillations were so smooth and quiet that one could not know or even hear that 5000 gallons of water were in oscillation. The elevation, acceleration, and all force traces were absolutely free from secondary oscillations so that no filters whatsoever were used between the transducer output and the recording equipment. Secondly, the method adopted enables one to cover all possible values of K for a given β and see the evolution of the forces over a period of about 30 minutes.

B. CIRCULAR CYLINDER MODELS

Seven circular cylinders with diameters ranging in size from 20.3 cm. (8 inches) to 5 cm. (2 inches) have been used in this study. The cylinders were turned on a lathe from aluminum pipes or plexiglass rods and polished to a mirror-shine surface. There is no doubt that the surfaces were hydrodynamically smooth. The length of each cylinder was such that it allowed 0.08 cm. (1/32 inch) gap between the tunnel wall and each end of the cylinder. As will be noted

later, the cylinder was prevented from moving towards one or the other wall by means of small O-rings attached to the round cantilever end of the force transducers. A double-ball precision bearing (SKF-2303-J) with an approximately 1.5 cm. (0.6 inches) bore was inserted at each end of the cylinder in aluminum housings which sealed the cylinder air tight. The other face of each bearing was flush with the end of the cylinder.

C. FORCE MEASUREMENTS

Two identical force transducers, one at each end of the cylinder, were used to measure the instantaneous in-line and transverse forces. The basic transducer was manufactured by B.L.H. Electronics, Inc. under the trade name LBP-1 and catalogue no. 420271. A photograph of the gage assembly is shown in figure 4. The gage had a capacity of 2224 N (500 lbf.) with an overload capacity of 200 percent. The deflection of the gage under 500 lbf. load was 0.25 mm. (0.01 inch). For the largest cylinder and amplitude encountered in the experiments, the maximum load was less than 200 lbf., and the deflection of the beam was less than 0.2 mm. (0.008 inches).

A special housing was built for each gage so that it can be mounted on the tunnel window and rotated to measure either the in-line or the transverse force alone. The bellows which protected the strain gages had to be waterproofed in such a manner that they would not adversely effect the operation

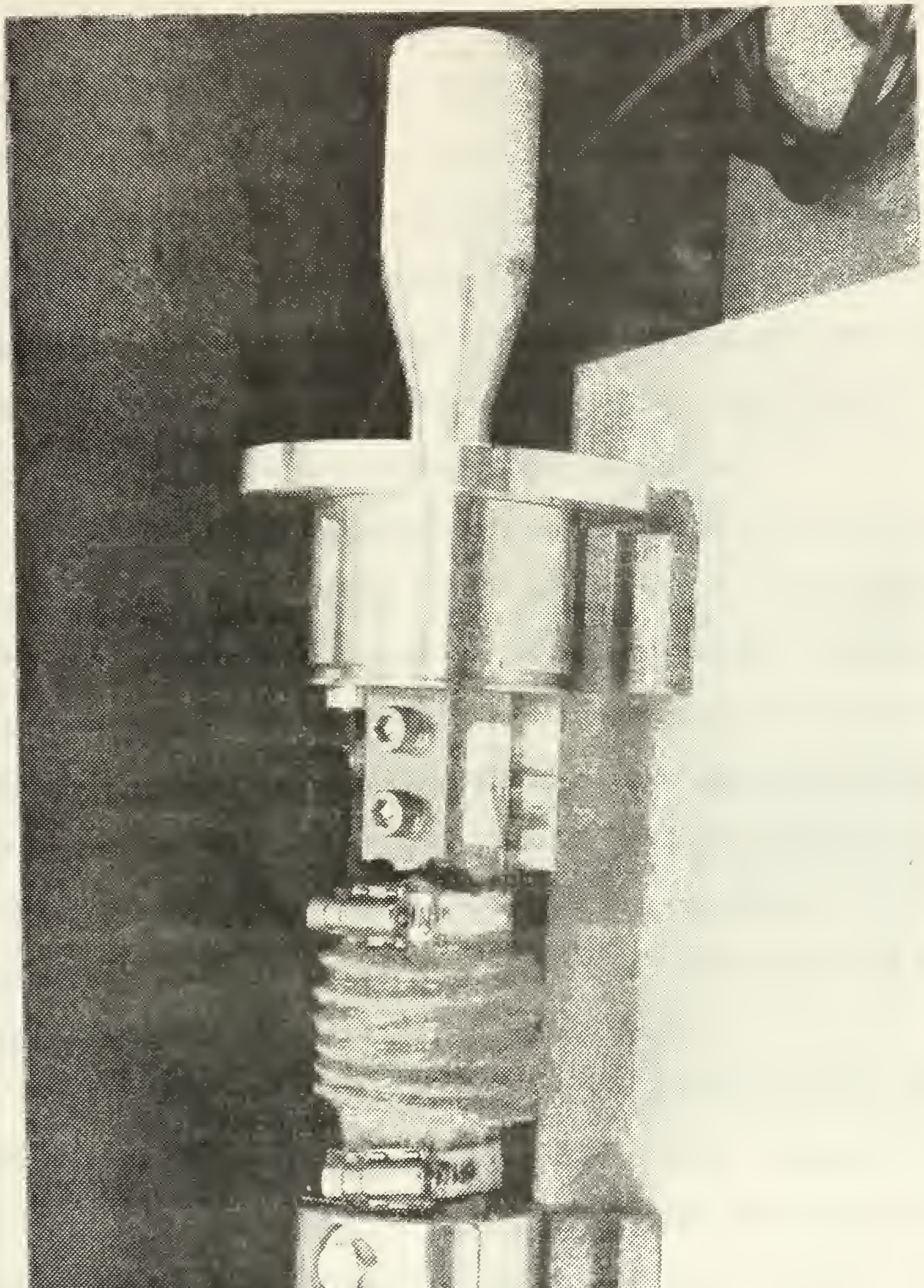

Figure 4. Gage Assembly

of the gages when subjected to about 6 m. (20 feet) water pressure at temperatures 18 C (64 F) to 74 C (165 F). For this purpose the bellows were completely filled with Dow Corning 3140 (RTV coating without bringing the rubber into contact with air during the filling operation. After filling, the ends of the bellows were sealed air tight with special clamps. The silicon rubber remained in its original liquid form throughout the operation of the gages.

The cylinders were placed in the test section by retracting the gages from their housing and then pushing them into the bearings mounted on each end of the cylinders. As noted earlier the O-rings placed on the cantilever end of each gage prevented the test cylinders from moving sideways towards one or the other wall and helped to exactly set the 0.8 mm. (1/32 inch) space between the cylinders and the tunnel wall. The cylinders were free to rotate at the application of a slight torque by hand.

After the mounting of the first cylinder, the exact angular position of the gages within their housing had to be determined and set with a pin so that the gages measure either only the in-line or the transverse force. For this purpose, an approximately 400 N load was hung on the cylinder with a lubricated nylon rope. The in-line force (acting in the horizontal direction) was observed on the amplifier recorder system. Then the gage was rotated in small increments until the in-line force was exactly zero. A final check was made by measuring the outputs of the gages with a precision

voltmeter. The position of each gage was marked and set with a pin. Finally, four bolts were placed on the gage housing to hold the gages rigidly in position. Removal of these bolts and the pin allowed the rotation of the gages exactly 90 degrees, after which the bolts and pin were placed in position. In this manner the gages were capable of measuring either the in-line or the transverse force without any "cross talk" between the in-line and transverse forces. Ordinarily, one gage was set to measure in-line force and the other gage the transverse force. At times both gages were used to measure only the in-line or the transverse force.

The calibration of each gage was accomplished by hanging loads in the middle of the cylinder after setting both gages to sense only the transverse (here vertical) force. The directional sensititivity of the gages was also checked by applying identical loads upwards on the test cylinders with the help of a hook-cantilever arm attached to the top of the tunnel outside the test section. Repeated calibrations have shown that the gages are perfectly linear up to 2000 N; they yeild the same signal for loads applied either downward or upwards; and that the gages, together with the electronic system to which they were eventually connected were capable of sensing forces as small as 0.1 N (0.02 lbf.).

The analog data appearing on a two channel Honeywell recorder in the form of either in-line force versus amplitude or transverse force versus amplitude were were read at every 0.1 seconds. The evaluation of the data is discussed later.

D. ACCELERATION, ELEVATION, OR VELOCITY MEASUREMENTS

It is because of the extreme importance of the accurate measurement of the instantaneous value of these quantities that they are discussed here separately.

Firstly, it should be noted that the measurement of the amplitude, acceleration, elevation, or the velocity is a matter of interpretation of the signal received from the appropriate transducer in light of one of the following expressions

$$U_m = \frac{2\pi A}{T}, \quad a_m = \frac{dU}{dt} = \left(\frac{2\pi}{T}\right)^2 A = \frac{2\pi}{T} U_m$$

in which $T = 5.272$ seconds for the experiments reported here.

Three transducers were used to generate three independent d.c. signals, each proportional to the instantaneous value of one of the quantities cited above. The first one consisted of a seven foot long platinum wire stretched vertically in one leg of the tunnel. The output of the capacitance-wire bridge was connected to an eight channel amplifier-recorder system. The response of the wire was found to be perfectly linear within the range of oscillations encountered. The wire was capable of yielding a measurable signal for changes in water elevation as small as 0.8 mm. (1/32 inches). Such a sensitivity was not, however, always desirable for the instabilities on the water surface gave rise to small oscillations in the analog records. The effect of such instabilities were practically eliminated by placing the wire along the axis of

30 cm. (1 foot) diameter and 213 cm. (7 feet) long thin plastic pipe. Be that as it may, the use of this wire was rendered unnecessary due to the use of a more reliable method.

The second method consisted of the measurement of the instantaneous acceleration by means of a differential-pressure transducer connected to two pressure taps placed horizontally 61 cm. (2 feet) apart and 91.5 cm. (3 feet) to one side of the test section. The output of the transducer was again connected to the eight channel recorder. The instantaneous acceleration was then calculated from $\Delta p = \rho s(dU/dt)$ where Δp is the differential pressure, s the distance between the pressure taps and dU/dt is the instantaneous acceleration of the fluid. The effect of the pressure drop due to the viscous forces over the distance s was calculated to be negligible.

The third method again consisted of the measurement of the differential pressure between two pressure taps. The two taps were placed symmetrically on the two vertical legs of the tunnel at an elevation 96.52 cm. (38 inches), (see figure 5) below the mean water level, i.e., $H = 38$ in.

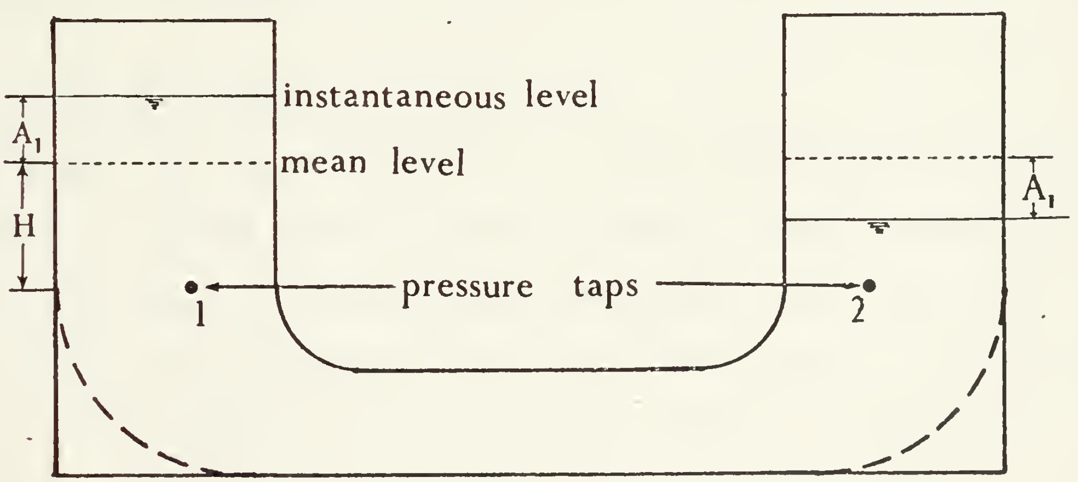

Figure 5

Applying Bernoulli's equation for unsteady flow between each pressure tap and the instantaneous level of water, it is easy to show that twice the amplitude of the free surface oscillation (virtual amplitude) is given by

$$A = 2A_1 = \frac{\Delta p/\gamma}{1 - \frac{1}{g}\left(\frac{2\pi}{T}\right)^2 H} \tag{10}$$

in which g and T are constant and H is kept constant. Thus the signal of this transducer yielded the virtual amplitude or the maximum velocity in each cycle. It was entirely free from noise or small free surface effect. The transducer was calibrated and its linearity checked before each series of experiments.

Suffice it to say that all three methods gave nearly identical results and yielded the amplitude, velocity, or acceleration to an accuracy of about two percent relative to each other. These comparisons as well as the perfectly sinusoidal and noise-free character of all pressure and force traces speak for the suitability of the unique test facility used in this study.

E. DATA REDUCTION

Experiments were repeated at least three times for each cylinder. Often only two, but at times as many as four runs were evaluated. Suitably selected cycles were read every 0.1 second and the tabulated data were transferred to the computer data cards together with additional relevant information such

as calibration factors, cylinder diameters, number of data cards, etc. A computer program calculated the force transfer coefficients previously discussed (see Appendix A).

Sample traces of the in-line force and the elevation are shown in figure 6.

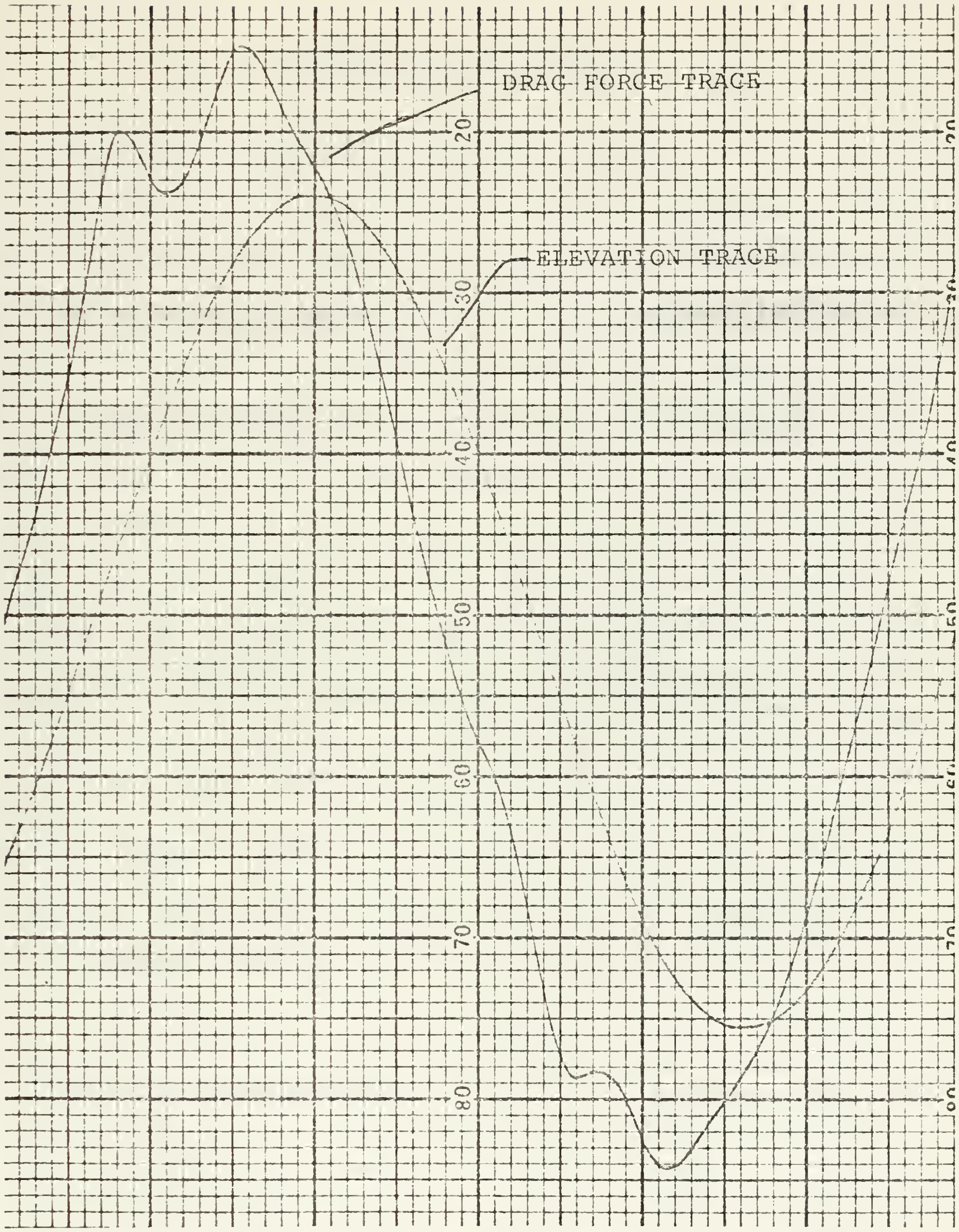

Figure 6. Elevation and Drag Force Traces

IV. DISCUSSION OF RESULTS

A. BLOCKAGE AND LENGTH-TO-DIAMETER EFFECTS

Attempts to achieve as high Reynolds numbers as possible in conducting wind-tunnel or water-tunnel experiments invariably give rise to wall-interference effects which, of course, influence whatever measurements are desired. There are several blockage correction formulas for steady flows which might be used so that the wall-interference effects might be minimized. Unfortunately none of these formulas could be used in the present study for no one has demonstrated that the blockage effects in oscillatory flow are identical to those experienced in steady flows.

The blockage ratio D/w , where D is the diameter of the cylinder and w the width or height of the test section, and the length- or span-to-diameter ratio, L/D , for the cylinders used in the present study are tabulated below (w = 91.44 cm. (3 feet), L = 90.885 cm. (2.9818 feet)).

D	D/w	L/D	$D^2/\nu T$
20.244 cm. (7.970 inches)	0.22	4.49	7968.7
16.447 cm. (6.475 inches)	0.18	5.52	5259.9
15.177 cm. (5.975 inches)	0.17	5.99	4480.2
12.674 cm. (4.990 inches)	0.14	7.17	3123.2
10.103 cm. (3.978 inches)	0.11	8.99	1985.2
7.544 cm. (2.970 inches)	0.082	12.05	1106.6
6.349 cm. (2.4996 inches)	0.069	14.31	783.8
5.057 cm. (1.991 inches)	0.055	17.97	497.2

* $\nu = 1.05 \times 10^{-5}$ ft^2/sec (0.009755 stokes)

Also shown in the above table are the $\beta = D^2/\nu T$ values for later use. Achenbach (7) and in some of the experiments of Fage and Falkner (8) the blockage ratios were 0.166 and 0.185 respectively. The length-to-diameter ratio in Fage and Warsap's (9) experiments was 20.2 or 7.88, depending on the diameter of the two cylinders they used, as compared to 3.33 in the experiments of Achenbach. It is generally observed that values of C_D are smaller for cylinders with larger length-to-diameter ratio. Thus the presence of both the gaps and the larger L/D ratios could result in lower drag coefficients. Furthermore, the wake of the cylinder, near the cylinder ends is supplied with high pressure fluid from the front and as a result smaller values of C_D are expected since the base pressure is increased over the value it would otherwise attain. In the subcritical range of Re, however, these effects appear to be negligible. In fact, Mosbach (10) found that in the subcritical range there is no effect of length-to-diameter ratio.

In the present experiments, the gap cannot be eliminated by extending the cylinder into a cylindrical cavity within the two windows supporting the gages and the cylinder because of the fact that eight cylinders of different diameters were used. Obviously, it would have been too costly to build a window for each cylinder. It is believed that the very small gap used plus the cantilever end of the gage extending into the bearing in the cylinder minimized the supply of high pressure fluid into the wake from the front of the cylinder during one half of the cycle.

Returning to the discussion of the blockage effect it must be emphasized that the formulas used for steady flow correction effects cannot be applied to oscillating flows and that there is not a unique blockage correction for the entire period of the harmonic flow. This is evident from the fact that within a given cycle the fluid undergoes varying accelerations and velocities as the wake width, momentum deficiency, and the wake pressure change accordingly. Thus, a blockage correction made for the instant of maximum velocity is not applicable to the instant at which the maximum acceleration occurs. In view of the fact that there are no previous investigations, a series of experiments had to be conducted to determine the role of blockage in the flow under consideration. For this purpose a differential pressure transducer was connected to two pressure taps on the same side of the tunnel wall. One of the taps was placed on the wall directly above the axis of the test cylinder. The other tap was placed 76 cm. (29.92 inches) to one side of the first tap along a line parallel to the flow. A series of experiments were carried out with the 16.447 cm. (6.475 inches) cylinder and the differential pressure was recorded and compared with the differential pressure obtained from the acceleration transducer. Furthermore, to simplify the comparison both transducers were calibrated so as to render exactly the same output under identical calibration loads. The results have shown that the two differential pressures were nearly identical and that they were certainly within three percent of each other.

Often the two traces of two transducers were undistinguishably coincident. This somewhat surprising result is a clear indication of the fact that the blockage effect in harmonic flows is negligible at least for D/w ratios less than 0.18. Although no special attempt was made to interpret the lack of blockage effect in such flows it is believed that the presence of vortices on both sides of the cylinder together with the high periods of acceleration and velocity render the flow relatively more uniform at short distances away from the cylinder in the test section. Therefore, for the reasons cited above no blockage-effect corrections were applied to the data presented here. It might be of interest to note that had the flow been assumed uniform and had the maximum velocity for the largest cylinder and the Reynolds number were used to calculate a blockage effect correction through the use of one of the existing formulas, one would have found that such a correction would have amounted to about six percent.

B. IN-LINE FORCE DATA

In-line force coefficients calculated as previously described are plotted as a function of the so-called Keulegan-Carpenter number denoted here by K. A representative plot of C_D versus K for the 16.447 cm. (6.475 inches) cylinder is shown in figure 7. It is evident from this figure that there is practically no scatter in the data even though it represents the result of four independent runs. This is

once again indication of the fact that the tunnel and its instrumentation are extremely reliable and that the data when evaluated with extreme regard relative to the phase angle are perfectly repeatable. As noted earlier the Reynolds number along this plot varies with K and β. One can designate special values of the Reynolds number along this curve simply by writing $Re = K\beta$ or by finding the K values which correspond to a particular value of Re by writing $K = Re/\beta$. Prior to doing so mean lines have been drawn through the data as carefully as possible for each cylinder data. Then round numbers such as 30,000, 40,000, etc. were chosen for Re and the corresponding K values were calculated and indicated on each plot. This simple procedure resulted in figure 8 in which C_D is plotted as a function of K for all cylinders and the identical Reynolds number points are suitably connected. This procedure has also been repeated for the inertia coefficient C_M and the result is presented in figure 9. It is evident from figures 8 and 9 that there is a remarkable correlation between the force coefficients, Keulegan-Carpenter number and the Reynolds number. The smoothness of the constant Reynolds number lines is another indication of the consistency of the data from one cylinder to another.

Figure 8 shows that for a given Reynolds number, for Reynolds numbers larger than approximately 50,000, C_D increases at first with increasing K, reaches a maximum, then decreases and then continues to rise gradually. For

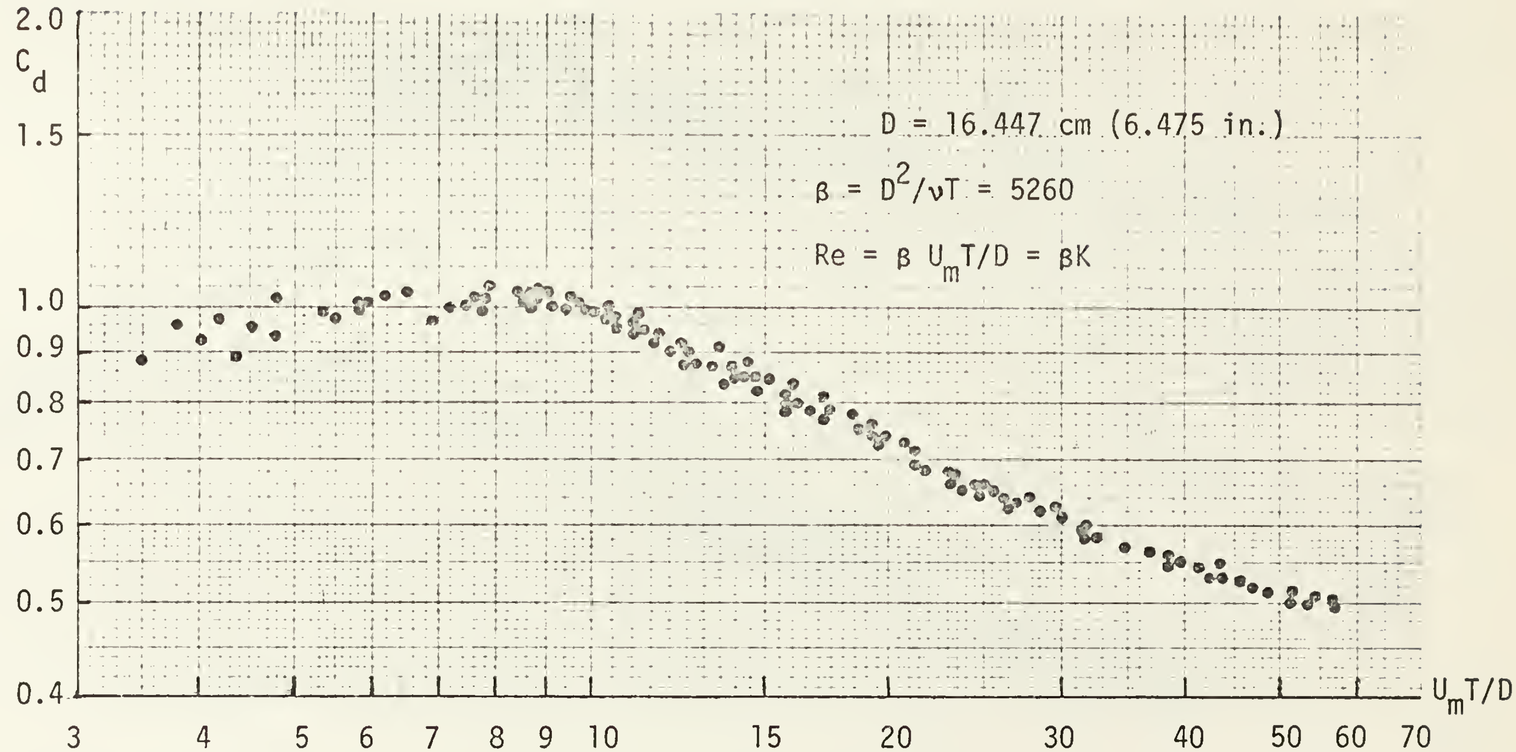

Figure 7. Drag Coefficient Versus Keulegan-Carpenter Number for D = 6.475 Inches Cylinder

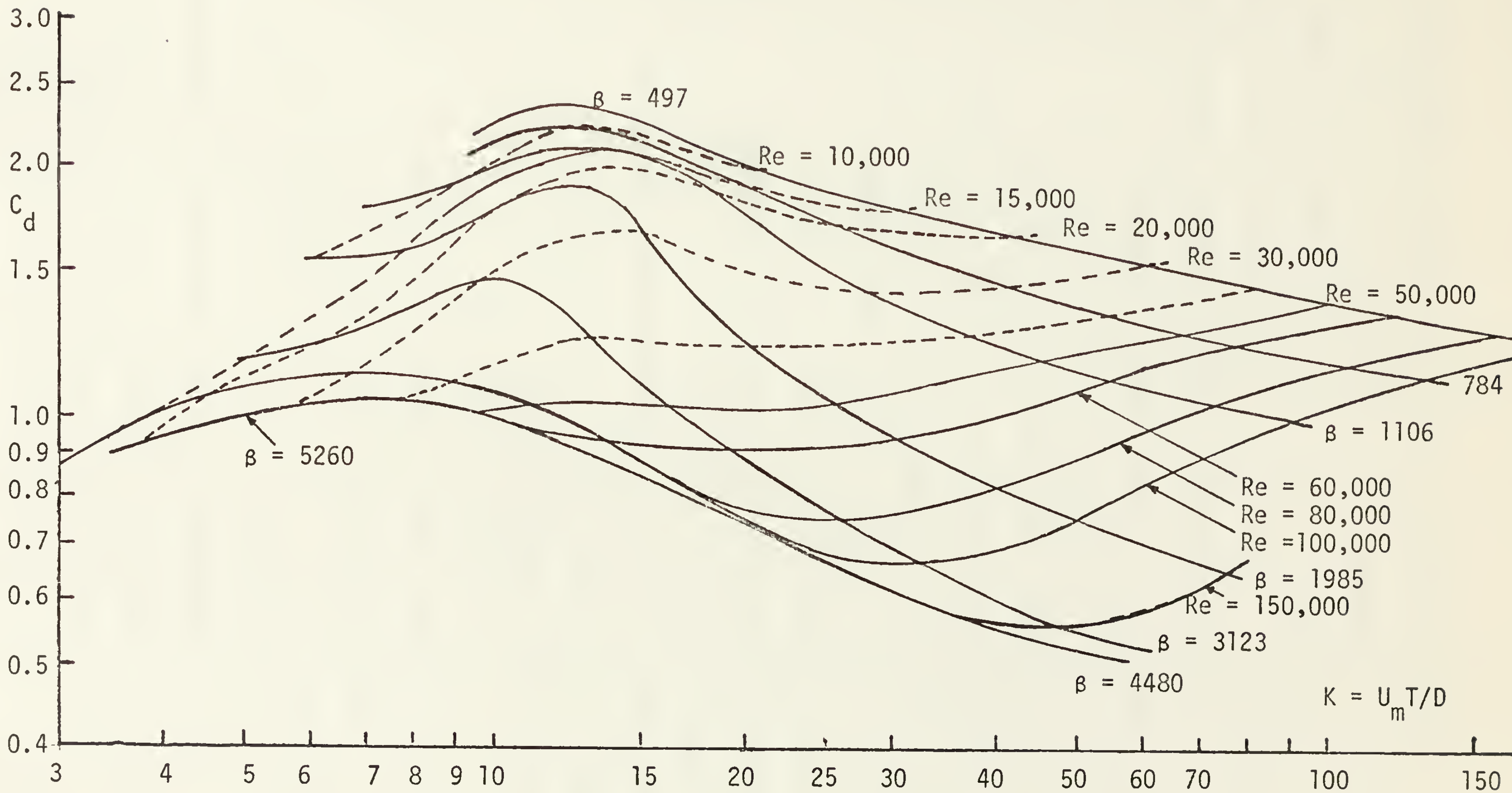

Figure 8. Drag Coefficient Versus Keulegan-Carpenter Number for All Test Cylinders

Reynolds numbers larger than about 50,000 the initial rise in C_D with K is very small or non existent. For very large Reynolds numbers and K, C_D attains a practically constant value of about 0.5. For a given K, however, C_D always decreases with increasing Reynolds number. It is evident from figure 8 that the change in C_D with Reynolds number for Re smaller than approximately 25,000 is almost negligible. It is for this reason that Keulegan and Carpenter (5) and Sarpkaya and Tuter (6) reached the conclusion that C_D does not depend on Reynolds number. The variations of C_D with Re become most apparent between Re = 25,000 and Re = 150,000. It is also noted that for large values of β such as $\beta = 5260$ and 4480 the C_D values do not significantly change with Re. In other words C_D values corresponding to $\beta > 5000$ yield the lower limit of the C_D values whereas C_D values corresponding to $\beta \simeq 500$ consitute the upper limit of the C_D values. The most significant changes in C_D with K and Re occur between these two β values.

Figure 9 shows the variation of the inertia coefficient C_M with K and Re. The most significant changes in C_M occur in the neighborhood of $K \simeq 12$. It is also around this value of K that C_M changes significantly with Re. As discussed in connection with the variation of the drag coefficient, β values of 500 and 5000 more or less determine the lower and the upper limits of the inertia coefficient. Figure 9 also shows that C_M for a given K increases with increasing Reynolds number and varies very

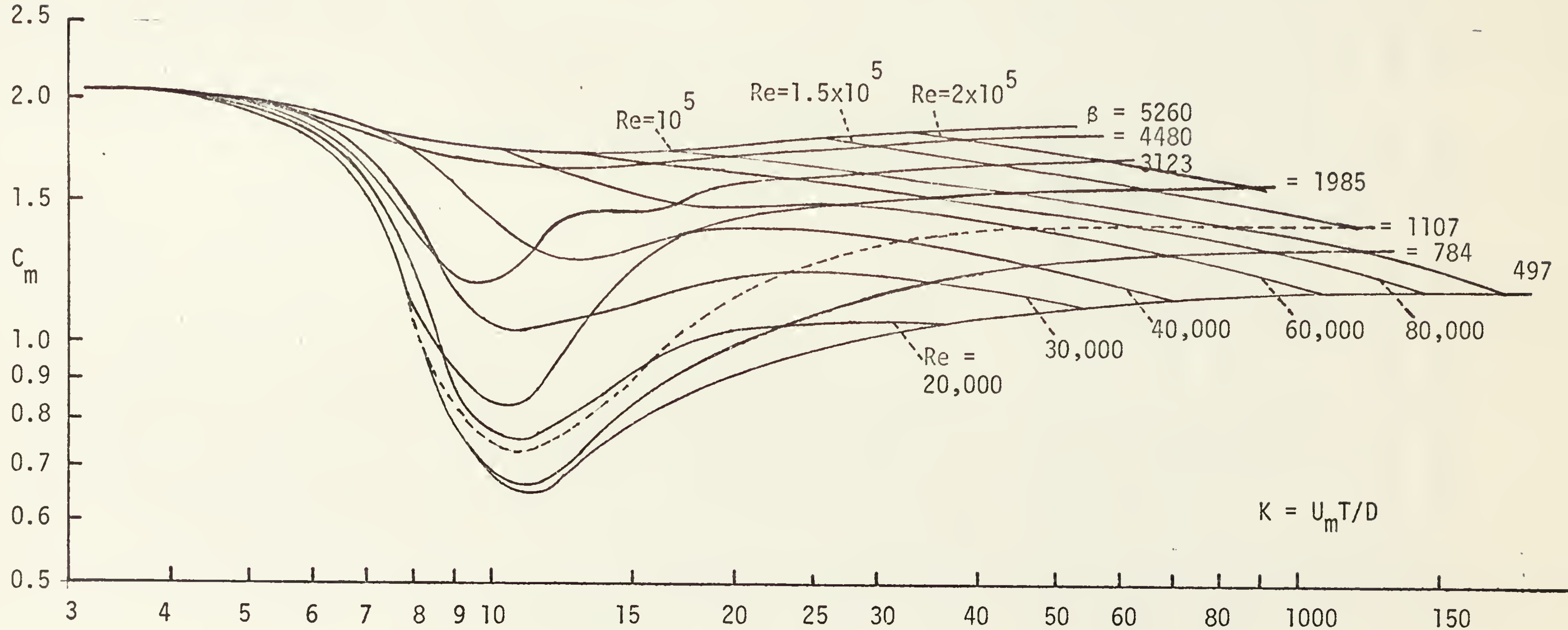

Figure 9. Inertia Coefficient Versus Keulegan-Carpenter Number for All Test Cylinders

little for Re smaller than approximately 25,000. It is for this reason that in the studies by Keulegan and Carpenter (5) and Sarpkaya and Tuter (6) C_M was thought to be independent of Reynolds number. The present results show that this is not so and that C_M varies with both Re and K throughout the K and Re range encountered in the present investigation. It appears that for very large Reynolds numbers C_M will approach a value of about 1.85.

V. CONCLUSIONS AND RECOMMENDATIONS

The present investigation of the in-line forces acting on smooth cylinders in the range of K values from about 3 to 200 and Re from 10,000 to 400,000 warranted the following conclusions.

1. The drag and inertia coefficients could be determined quite accurately through the use of the present experimental system.
2. Both the drag and inertia coefficients depend on the Reynolds and Keulegan-Carpenter numbers. For a given K, the drag coefficient decreases with increasing Reynolds number whereas the inertia coefficient increases with increasing Reynolds number.
3. The most significant changes in both C_D and C_M occur in the neighborhood of $K = 12$. β values of about 500 and 5000 determine the upper and lower limits of the two coefficients.
4. Neither C_D nor C_M vary appreciably with Re for $Re < 25,000$.
5. It is recommended that the experiments be carried out at even larger Reynolds numbers through the use of the existing systems by decreasing the viscosity of the water through heating and with cylinders of various artificial roughnesses.

6. It is also recommended that additional investigations such as the hydroelastic oscillations of flexibly mounted cylinders in harmonic flow and the hydro-elastic oscillations of ridigly mounted flexible cylinders be carried out. Finally the determination of the effect of wall proximity on the in-line and transverse forces acting on cylinders and other bluff bodies will be of extreme importance.

APPENDIX A

COMPUTER PROGRAMS

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                    C
C      AVERAGE CD AND CM CALCULATIONS FOR CYLINDERS                  C
C                                                                    C
C      TIME=DIMENSIONLESS TIME (TIME/PERIOD)                         C
C      BETA=DIMENSIONLESS DISPLACEMENT (UMAX*PER/DIA)
C      AMP=AMPLITUDE OF MOTION IN FEET                               C
C      ZI=FORCE COEFFICIENTS                                         C
C      CM=INERTIA COEFFICIENT                                        C
C      PER=PERIOD IN SECONDS (CHART PERIOD/CHART SPEED)              C
C      CD=DRAG COEFFICIENT                                           C
C      REMF=REMAINDER FUNCTION                                       C
C      AI,BI=FOURIER COEFFICIENTS                                    C
C      N=NUMBER OF DATA SETS                                         C
C      CN=CONVERTION FACTOR                                          C
C                                                                    C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
      DIMENSION FORCE(36)
      N=12
      DO 100 I=1,N
      G=32.174
      READ(5,55) (FORCE(K), K=1,36)
      RHO=62.4/G
      PI=3.14159
      CNU=0.000000105
      CL=1.49+6
      READ(5,10)DIA,AMP,PER,NCARD,UMX,CN,FMMM,NRUN
      Z1=(2*PER*PER)/(PI*PI*PI*DIA*DIA*CL*RHO*AMP)
      Z2=(-3*PER*PER)/(8*RHO*DIA*PI*CL*AMP*AMP)
      Z2L=(-4*PER*PER)/(3*PI*PI*RHO*DIA*CL*AMP*AMP)
C     COMPUTE BETA
      BETA=2*PI*AMP/DIA
      REYNO=(UMX*DIA)/CNU
      WRITE(6,15)DIA,AMP,PER,UMX,CN,NRUN
      WRITE(6,20)
      TIME=0.0
      SUM=0.0
      DO 65 K=1,NCARD
      SUM=SUM+FORCE(K)
65    CONTINUE
      FMEAN=SUM/NCARD
      DO 85 K=1,NCARD
      FORCE(K)=FORCE(K)-FMEAN
85    CONTINUE
      FMMM=FMMM-FMEAN
      CM=0.0
      CD=0.0
      CMLS=0
      CDLS=0
      FAA=0.0
      FBB=0.0
      FCC=0.0
      FDD=0.0
      FEE=0.0
      DELTAT=0.027972028
      DO 200 K=1, NCARD
      F=CN*FORCE(K)
      ALPHA=2*PI*TIME
      SINA=SIN(ALPHA)
      COSA=COS(ALPHA)
      FSINA=DELTAT*F*SINA
      FCOSA=DELTAT*F*COSA
      FLS=DELTAT*F*COSA*ABS(COSA)
      FAAS=2*PI*F*DELTAT*COSA*COSA*COSA*COSA*F
      FBBS=2*PI*F*F*COSA*ABS(COSA)*DELTAT*F
```

```
      FCCS=2*PI*F*SINA*COSA*ABS(COSA)*DELTAT*F
      FDDS=2*PI*F*SINA*SINA*DELTAT*F
      FEES=2*PI*F*F*SINA*DELTAT*F
      FAA=FAAS+FAA
      FBB=FBBS+FBB
      FCC=FCCS+FCC
      FDD=FDDS+FDD
      FEE=FEES+FEE
      CM=FSINA+CM
      CD=FCOSA+CD
      CMLS=FSINA +CMLS
      CDLS=FLS+CDLS
      WRITE(6,30)TIME,ALPHA,COSA,SINA,F,FCOSA,FSINA
      TIME=TIME+DELTAT
  200 CONTINUE
      CM=Z1*CM
      CD=Z2*CD
      CMLS=Z1*CMLS
      CDLS=Z2LS*CDLS
      CMFF=(Z1/2.0)*(FEE*FAA-FCC*FBB)/(FDD*FAA-FCC*FCC)
      CDFF=(-4.0*Z2/(3.0*PI))*(FEE*FCC-FDD*FBB)/(FDD*FAA-FCC*FCC)
      WRITE(6,35)
      WRITE(6,40)CM,CD,BETA,REYNO,CMLS,CDLS,CMFF,CDFF
      ANGLE=0.0
      WRITE(6,45)
      TIME=0.0
      DO 300 K=1,NCARD
      F=CN*FORCE(K)
      THETA1=((2.0*PI)/360)*ANGLE
      C1=(ABS(COS(THETA1)))*COS(THETA1)
      C2=RHO*((UMX**2.0)/2.0)*DIA*CL
      C3=((PI**2)*DIA*SIN(THETA1))/(UMX*PER)
      F1=(CM*C3-CD*C1)
      FLS=(CMLS*C3-CDLS*C1)
      FFOR=(CMFF*C3-CDFF*C1)
      F=F/C2
      REMF=ABS(F)-ABS(F1)
      FMAX=FMMM*CN/C2
      REMF=REMF/FMAX
      RLS=(ABS(F)-ABS(FLS))/FMAX
      RFOR=(ABS(F)-ABS(FFOR))/FMAX
      WRITE(6,50)TIME,F,F1,REMF,FLS,RLS,FFOR,RFOR
      ANGLE=ANGLE+10.06993
      TIME=TIME+DELTAT
  300 CONTINUE
  100 CONTINUE
   10 FORMAT(3F8.4,I8,3F8.4,I8)
   15 FORMAT('1',2X,'DIA=',F8.4,2X,'AMP=',F8.4,2X,'PER=',
     1F8.4,2X,'UMX=',F8.4,2X,'CN=',F8.4,2X,'NRUN=',I6)

   20 FORMAT('0',3X,'TIME/PER',7X,'ALPHA',7X,'COSA',15X,'SINA
     2',2X,'F',7X,'FCOSA',5X,'FSINA')
   30 FORMAT(' ',3F12.4,F19.4,3F12.4)
   35 FORMAT('0',7X,'CM=',7X,'CD=',7X,'BETA=',7X,'REYNO=',7X,
     1'CMLS=',7X,'CDLS=',7X,'CMFF=',7X,'CDFF=')
   40 FORMAT('0',6F12.4,2F12.4)
   45 FORMAT('0',7X,'TIME',9X,'F',9X,'F1',9X,'REMF',9X,'FLS',9X,
     1'RLS',12X,'FFOR',12X,'RFOR')
   50 FORMAT('0',6F12.4,2F15.4)
   55 FORMAT(F10.4)
      STOP
      END
```

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